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THE EFFECT OF THE SHAPE OF THE WATER-FLOW SYSTEM ON THE CALCULATION OF POWER PARAMETERS

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THE EFFECT OF THE SHAPE OF THE WATER-FLOW SYSTEM ON THE CALCULATION OF POWER PARAMETERS

I. M. Chorniy

Translation of "Pro vpliv formi protochnoy chastini vodometa
na rozrakhunkovi parametri rushiynogo kompleksu"
Visti institutu Gidrologiy i gidrotekhniki,
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THE EFFECT OF THE SHAPE OF THE WATER-FLOW SYSTEM ON THE CALCULATION OF POWER PARAMETERS

I. M. Chorniy

ABSTRACT

Modern methods of designing water-pressure engines incorporate the characteristics of axial flow pumps or performance curves of insulated propellers. The disadvantage of these methods is the lack of reliable data on the interaction between the engine and the hull of the fireboat, which are determined experimentally. This article cites data characterizing the interaction between the engine and the hull obtained in tests with a large-scale model of a double-shaft fireboat with different shapes of the suction duct. It has been shown that this form has a substantial effect on the propulsive capacity of the power complex.

The basic experimental relations are presented in dimensionless magnitudes in the form of diagrams that can be used in planning and designing water-pressure engines by the "equivalent propeller" method.

The mass production of fireboats in our country began over a decade ago, but reliable methods of designing them are still unavailable. /69

At the same time, the designers have insufficient experimental data on the performance of the water-squirting system at their disposal. Such data have been obtained primarily in independent tests with one or two different boat models.

The designs of the impellers in almost all proposed fireboats were based on the definition of the axial flow pump parameters.

Such calculations were used by the Kiev Planning and Design Bureau, the Leningrad Central Design Bureau, the Krasnoyarsk shipyard, etc.

We shall dwell briefly on the methods of calculating the initial parameters of the recently proposed fire pumps (water engines).

N. V. Koritov (ref. 1) recently developed a method in which the impeller is selected according to a diagram of the optimal values of pressure H (m) and consumption Q (m^3/sec) for certain series of axial wheels.

These values, in turn are defined on the basis of the known prop P and the speed of the boat v.

The initial formulas for defining these parameters are

$$K_Q = \frac{Q}{\overline{Q}} = \frac{P}{\rho v(a-w)} \cdot \frac{1}{\overline{Q}}, \quad (1)$$

$$K_H = \frac{\gamma H}{\overline{H}} = \frac{\gamma}{\overline{H}}(ka^2 - \beta)\frac{v^2}{2g}. \quad (2)$$

Here K_Q and K_H are the water consumption and pressure factors, which are used to find the optimal value of diameter D and number of revolutions n of the impeller from a general-purpose diagram; $\overline{H}$ and $\overline{Q}$ are the known dimensionless parameters which meet the optimal working conditions of a given type of wheel; $a = \frac{v_s}{v}$ are the required speed factors; k, w and β are the coefficients of the interaction between the hull and the engine. These coefficients can only be determined experimentally.

Inasmuch as the coefficients are unknown, it is impossible to make use of this method. /70

It is also not clear how to determine the required prop of the engine, whose resistance to the ship's hull must be taken into account.

Another method of calculating the design of fireboats, based on the "equivalent propeller" system and involving the use of curves of insulated propeller performance, has been proposed by the Leningrad Institute of Water Transport (LIWT) (ref. 2).

The equivalent propeller parameters K_p and λ_p are in this case defined by the following formulas

$$K_p = \frac{P_e}{\rho n^2 D^4 (1+t)}, \tag{3}$$

$$\lambda_p = \frac{v_p}{nD} = \frac{v}{nD}\left(\bar{v}_s - \frac{\sigma_e}{4(1+t)\bar{v}_s}\right), \tag{4}$$

where t is the suction coefficient of the fireboat screw operating in a pipe outside the hull, as defined in the experiment

$$t = \frac{P_e - P}{P} = -\frac{P - P_e}{P}, \tag{5}$$

P_e and P are the effective power of the complex and the prop of the propeller, respectively; $\bar{v}_s = \frac{v_s}{v}$ is the unrated water flow through the fireboat propeller; v_s is the rate of the water flow through the fireboat propeller.

Charts with experimental curves t and $\bar{v}_s$ denoting the load factor of the entire complex are cited.

$$\sigma_e = \frac{P_e}{\frac{1}{2}\rho F v^2}. \tag{6}$$

Using the theory of interaction between a conventional engine and the ship's hull (ref. 3), A. M. Basin proposes an initial equation in a somewhat different form (ref. 4).

The coefficient of the indicated rate of flow to the propeller disk, as it operates in pipe k_w, is therefore introduced

$$v_s = v_e + k_w w_a. \tag{7}$$

The influence factor of the water-discharge pipe ζ_T is also introduced. We can then find the expression for the load factor of a conventional engine

$$\sigma_k = \frac{2P_k}{\rho F v_e^2} = \frac{\sigma_e + \frac{\zeta_\tau}{2}(\sqrt{1+2k_w\sigma_e}+1)}{1+k_w\frac{\zeta_\tau}{2}}, \tag{8}$$

as well as for the suction factor t_b and calculated speed v_p

$$t_b = f(k_w, \sigma_k), \quad v_p = f(k_w, \sigma_k, t_b). \tag{9}$$

Formulas (6), (7), (8) and (9) contain the following designations: w_a is the rated speed far above the engine speed; $v_e = v(1 - \psi_b)$ is the ship's speed in relation to the surrounding water; ψ_b is the coefficient of the following current; σ_e is the complex load factor of the useful haulage attributed to speed v_e; P_k is the overall haulage of a conventional power installation.

The first approximation includes the following experimental coefficients /71

$$\varphi_b = 0.2; \quad k_w = 0.66; \quad \zeta_\tau = 0.3\text{--}0.35.$$

Although these data were obtained in the investigations of actual ships and models, they apply to only two types of available boats and cannot be considered universal.

Thus, the relations $t = f(\sigma_e)$ and $\bar{v}_s = f(\sigma_p)$ obtained by S. P. Medvedev (ref. 5) apply to the form of channel characterizing the single-jet fireboats among the serial tug boats 861 and 878.

The interaction between the engine and the hull of a two-jet fireboat (ref. 6) has been defined by us with reference to a self-propelled model with a suction portion of the water duct, as outlined in figure 1c.

We believe that the interaction factors depend in large measure on the design characteristics of the hull and water duct. Thus, in order

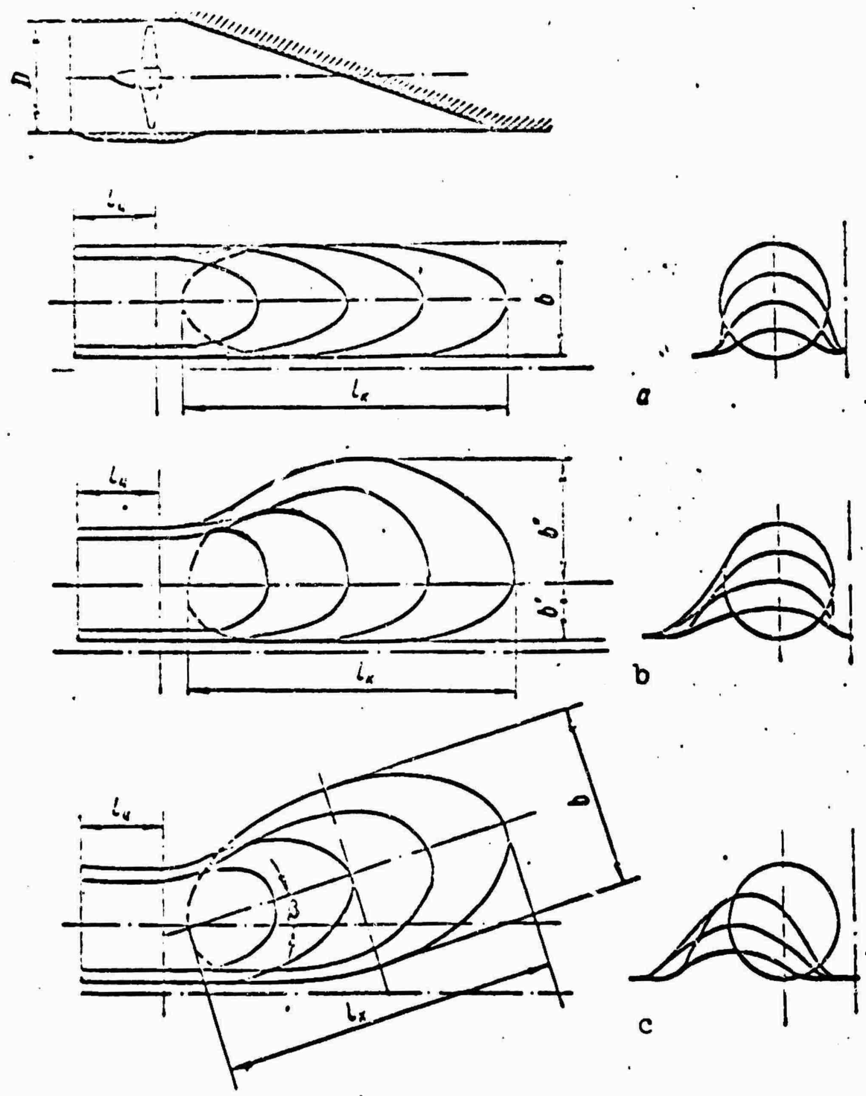

Figure 1. Possible shapes of suction part of water ducts with bottom water-intake. a, direct intake without transversal opening ($b/D = 1$); b, direct intake from asymmetrical opening ($b' \neq b''$); c, indirect intake ($\beta > 0$).

to make effective use of the method proposed by the Leningrad Institute of Water Transport, we must find the data on the effect of the shape of the duct on these parameters. This will improve not only the calculations connected with the selection of a propeller, but also establish a better alternate design, that may lower the cost of reciprocal action. /72

The usual and possible shapes of the water ducts, characterized by a bottom intake and the ejection of the straight jet, are shown in figure 1.

We will introduce certain geometric parameters to describe the design characteristics of the water duct in the first approximation

b/D - cross section--the ratio between the largest lateral axis of the inlet contour and the diameter of the cylindrical part of the pipe;

l_k/D - longitudinal section--the ratio between the longitudinal axis of the inlet pipe socket and the diameter of the pipe;

β - the angle between the longitudinal axis of the inlet pipe socket and the axia of the cylindrical part of the pipe;

l_c/D - the relative length of the outlet portion of the pipe.

Usually these parameters cannot be used to provide a full description of the geometry of the outlet portion, but they give a fair idea of its shape. The l_k/D value, for example, indicates the listing of the underwater lines at the stern, and together with b/D it reveals the contour line and its area (in the first approximation, the contour line may be considered elliptical in shape).

To characterize the scheme presented in figure 1b, the asymmetrical cross section, the following parameters should be added: b'/D and b''/D. If $\beta = 0$, it is obvious that $b' = 1/2\ D$; $b'' = b - 0.5D$.

To determine the effect produced by the shape of the channel on the interaction, a special metal model was made of the stern, which made it possible to change the shape of the suction channel within a fairly wide range.

The tested self-propelled models with the new type of stern were equipped with the kind of channels shown in figure 1a. Their cross section was $b/D = 1$, and the longitudinal section l_k/D changed within the range of 3.9-11.5. The effect of the listing channel arch on the nature of the interaction, with $b/D = 1$, has been ascertained.

The relative length of the cylindrical part remained a constant $l_c/D = 1.5$.

The suction coefficient for various values σ_e was determined from the action curves of the complex outside the boat, plotted by the use of the experimental data and by the previously described method (ref. 6).

Figure 2 shows a comparison of the curves representing the changing coefficients of prop K_p and moment K_m depending on the reciprocal load factor, based on the effective haulage, for the three l_k/D values.

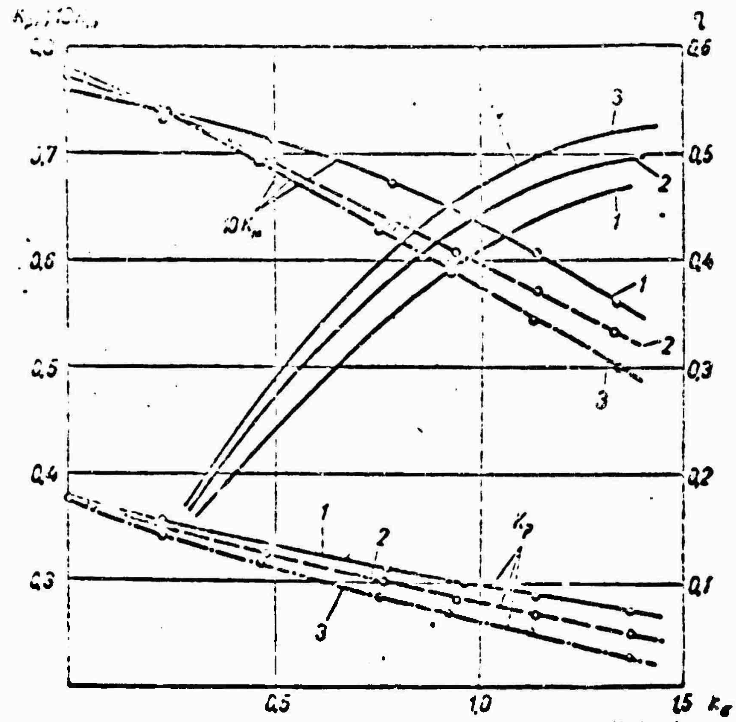

Figure 2. Effect of incline of underwater suction line K_p and K_m factors (for model shown in fig. 1a with b_k/D = 3.9-11.4): 1, l_k/D = 3.9;. 2, l_k/D = 6.5; 3, l_k/D = 11.4.

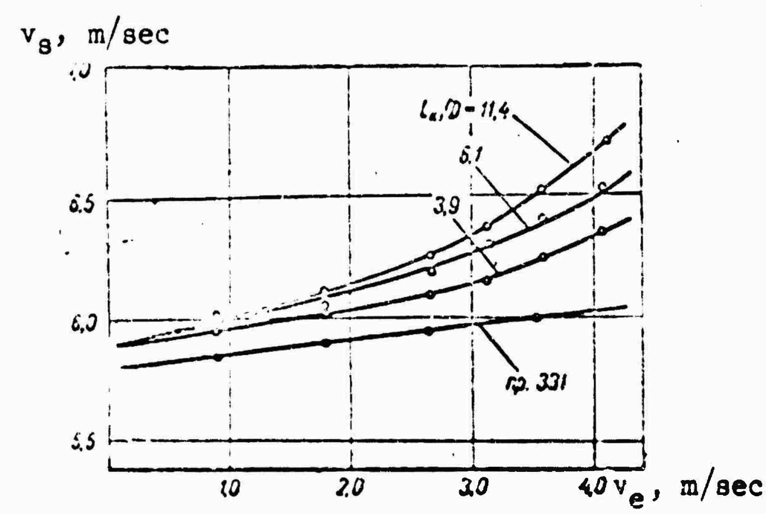

Figure 3. Effect of incline of underwater suction line on rate of water flow through propeller disk v_s.

The relationship between the K_d value and the load factor is

$$K_d = \frac{\lambda}{\sqrt{K_e}} = \frac{1}{\sqrt{\frac{\pi}{8}\sigma_e}}.$$

Inasmuch as the curves representing the hull support remain almost unchanged in all three cases, and the action curves apply to the same number of the propeller revolutions per second, it may be assumed that the comparison is based on λ-idem, that is v-idem.

In these conditions, the decrease in the state of rest and momentum from the high l_k/D level is indicative of a higher propeller pitch, which is made possible by an increase in the rate of v_s.

This is confirmed by direct measurements of the flow rate v_s by the /73
installation of a special butterfly in the water duct outside the propeller. The curves $v_s = f(v_e)$ for all the three cases are presented in figure 3, and here $v_e = v(1 - \psi)$; $\psi = 0.1$.

An increase in speed v_s, with the speed of movement v and σ_e remaining the same, results in a higher engine efficiency, as may be seen from the following formula

$$\eta_i = \frac{1}{1 + \frac{1}{4}\sigma_e \frac{v}{v_s}}.$$

However, if an increase in l_k/D reduces the rest factor K_p, with /74
the hull-support curve $R = f(v)$ remaining almost invariable (and the comparison based on v-idem), the suction factor t should be reduced.

In figure 4 we show the relationship $t = f(l_k/D)$ for various lead factors σ_e. With $b/D = 1$, magnitude l_k/D changes within the range of 3.9 to 11.4, which should be considered as most likely in the case of actual boats.

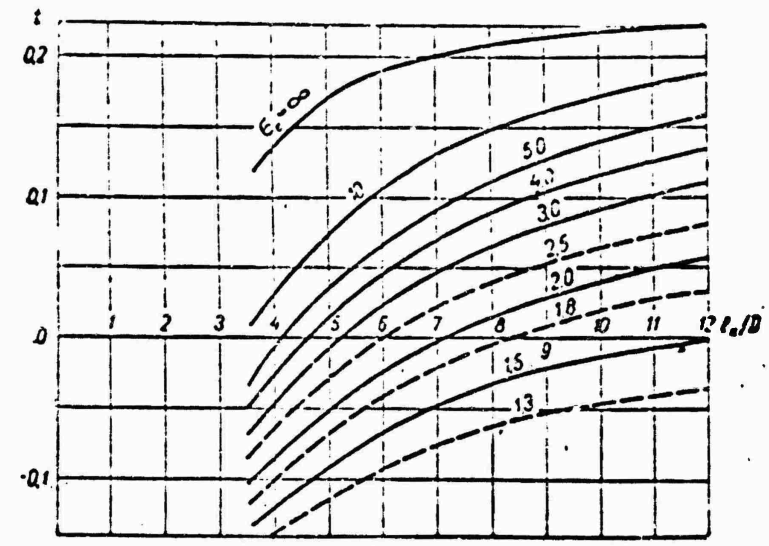

Figure 4. Diagram for determining suction factor $t = f(\sigma_e)$ which takes into account shape of duct shown in figure 1a ($h/T > 7$; $T_B/D = 1$).

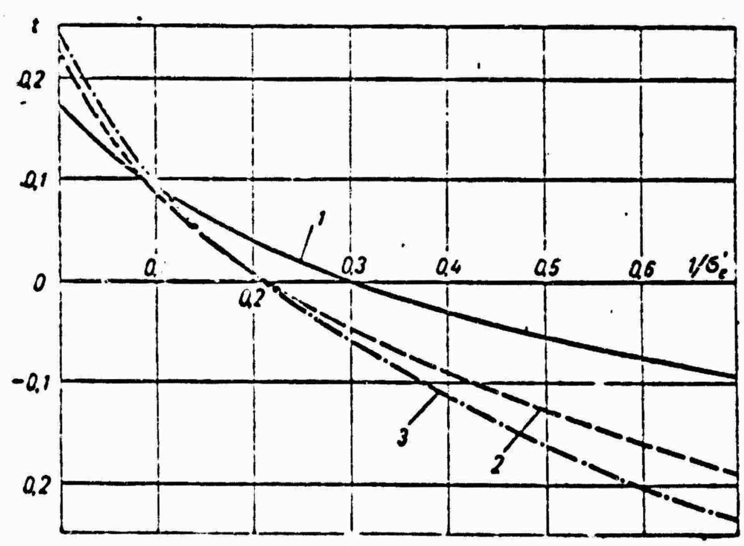

Figure 5. Effect of transversal opening b/D and angle β on suction factor t($h/T > 7$; $T_B/D = 1$). 1, $l_k/D = 5.5$; $b/D = 1$; $\beta = 0$; 2, $l_k/D = 5.7$; $b/D \approx 2.4$; $\beta \approx 12^o$; 3, $l_k/D \approx 8.0$; $b/D \approx 3.1$; $\beta \approx 18^o$.

A diagram could also be drawn up for calculations applying to deep water in which a stream is ejected underwater, and the ejection nozzle is cylindrical in shape.

Figure 4 shows that, with $l_k/D \approx 11$ for the values σ_e representing the rated "purely self-propelled" movement (σ , 1.5-2.0), the value t is equal to 0 to 0.5, while in the same conditions characterizing a diesel-powered ship 331, $t \cong -0.2$. /75

The propulsion factor for $l_k/D = 11$ equals $\eta_p = 0.5$ and for the diesel-powered model 331 it is $\eta_p = 0.44$ (underwater ejection). The propulsion coefficient has been increased by $\frac{\Delta\eta_p}{\eta_p} \approx 14$ percent, primarily by reducing the suction losses.

The effect produced on coefficient t by an increase in the cross section and the curvature of channel axis may be judged from figure 5. Curve 3 represents the variant suction portions of the channel in the diesel boat 331.

Thus, in the case of fast fireboats (in small roadsteads), the "direct inlet" and small opening of the suction part of the channel should be considered as more expedient. In the case of tugboats, it is possible to use a considerable transversal opening ($l_k/D = 2.5$-3.5), which increases the positive suction under conditions similar to mooring conditions.

Inasmuch as speed v_s, assuming the same σ_e, also changes with the changing shape of the channel, it was possible to find the coefficient k_w by the use of the investigation data. Due to the installation of an aiming device outside the propeller to straighten the stream of water, and the direct calculation of support P on the basis of the following formula

$$P = \rho F w_a\left(v_e + \frac{w_a}{2}\right),$$

we get

$$\frac{w_a}{v_e} = \sqrt{1+\sigma_p} - 1. \tag{10}$$

Further, we find from (6)

$$k_w = \frac{v_s - v_e}{w_a}. \tag{11}$$

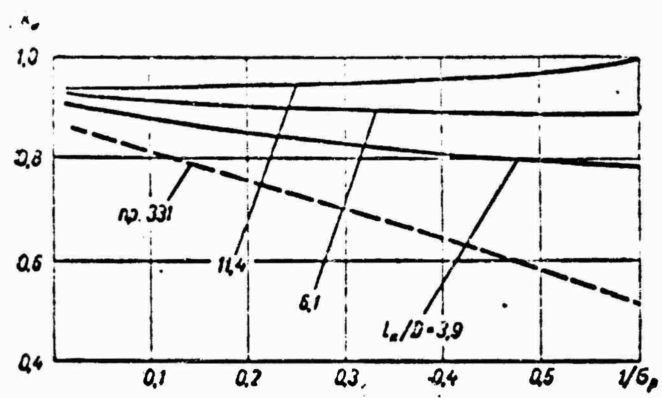

Figure 6. The $k_w = f(\sigma_p)$ relation of incline of underwater lines outlined in figure 1a.

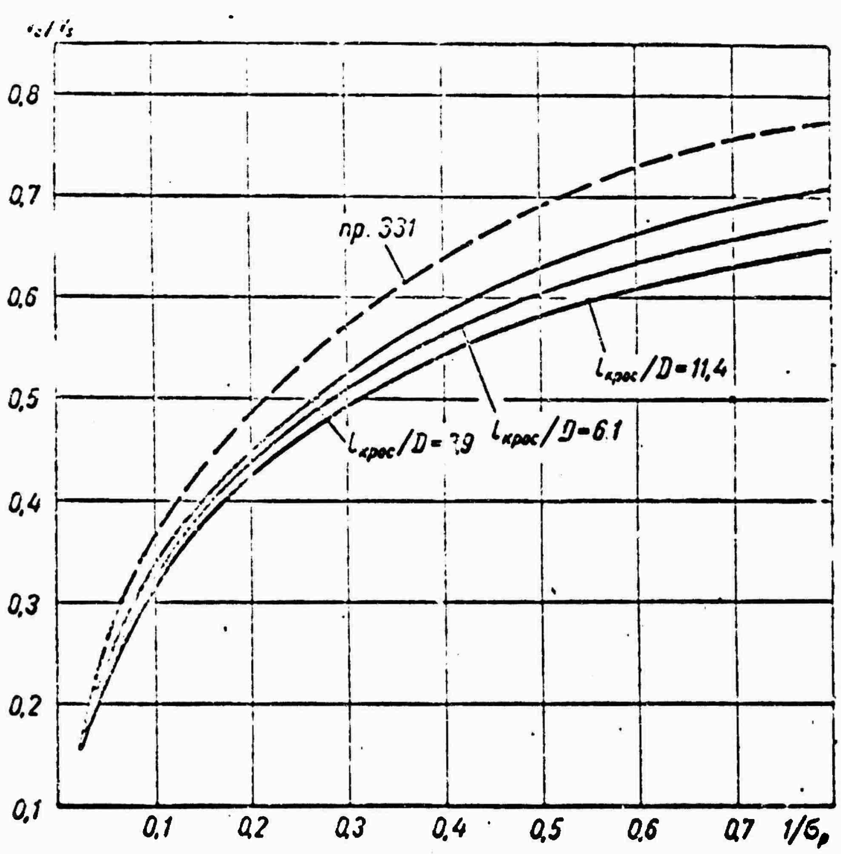

Figure 7. Diagram for determining rate of water flow through propeller disk v_s ($T_B/D = 1$; $h/T > 7$; $\psi = 0.1$).

The relationship $k_w = f(\sigma_e)$ is plotted in figure 6.

That figure shows that k_w changes perceptibly with the change in σ_e.

The diagram of the $v/v_s = f(1/\sigma_p)$ dependence, which takes into account the construction characteristics of the suction part of the channel (fig. 7), can be used for the calculations in the initial stage.

This diagram applies also to the relative depth of the propeller $T_B/D = 1$-0.5). /76

The dependence curves of $t = f(l_k/D, \sigma_e)$, which can be determined from figure 4, are presented in case $T_B/D \geq 1$.

If $T_B/D < 1$, a more precise definition of the suction factor can be achieved by the use of the following formula

$$1+t' = \frac{1+t}{1+\frac{\gamma F h_\pi (1+t)}{P_e}}. \qquad (12)$$

As revealed by the test, the support coefficient

$$K_p = \frac{P}{\rho n^2 D^4}$$

is not affected by the changing number of propeller revolutions except when it reaches full depth, that is, when $T_B/D = 1$.

With $T_B/D < 1$, the reduced number of revolutions increases the K_p value, and the smaller T_B/D, the greater the increase.

This can be seen from the following formula

$$K_p' = K_p + \frac{\pi g h_\pi}{4 n^2 D^2}, \qquad (13)$$

where K_p is the support coefficient found with $T_B/D = 1$; h_π is the constant not dependent on the engine operation, but only on the T_B/D

magnitudes, and it represents the center of gravity of the stream intersection above the water surface. /77

It follows from this that the graph $t = f(T_B/D, \sigma_e)$ drawn up experimentally for a certain number of motor revolutions per second, will be somewhat changed by a different number of revolutions.

We believe that the following method should be used to find a more precise t value, following the preliminary calculation of the engine by the method of the Leningrad Institute of Water Transport.

Given some values $D_1, D_2, \ldots, D_i, \ldots$, the known magnitude P_e can be used to define the load factors $\sigma_{e_1}, \ldots, \sigma_{e_2}, \ldots, \sigma_{e_i}, \ldots$ and, therefore also,

$$\left(\frac{T_B}{D}\right)_1, \ldots, \left(\frac{T_B}{D}\right)_2, \ldots, \left(\frac{T_B}{D}\right)_i, \ldots$$

given the draught of the boat.

Coefficient t is determined for each σ_e value from the graph of figure 4, which corresponds to the case of $T_B/D = 1$.

Further, according to reference 6, the h_π magnitude is found for each $(T_B/D)_1$ and $(T_B/D)_2$ value.

By using formula (12), it is possible to calculate $t_1', \ldots, t_2', \ldots, t_i', \ldots$, which correspond to a partial depth.

We shall now discuss the loss factor in pipe ζ_T.

As is known, the effective power of the entire boat, i.e., the engine working in ideal surroundings (water), is determined by the following formula

$$P_k = \rho F v_s w_a. \qquad (14)$$

Then

$$\frac{P_K - P}{P} = t_a \tag{15}$$

will represent the suction factor of the entire complex under investigation in an ideal liquid.

Here P is the propeller support which, in the first approximation, can be taken as equal to the support found by a direct measurement of the model.

However, the magnitude

$$t = \frac{P_e - P}{P} \tag{16}$$

represents the suction coefficient of the actual complex, if the effective hauling power P_e is equal to the hull support.

A. M. Basin (ref. 4) finds the following relationship between P_k and P_e, on the one hand, and coefficient ζ_T, on the other

$$P_e = P_K - R_T, \tag{17}$$

where

$$R_T = \zeta_T \rho \frac{v_s^2}{2} F. \tag{18}$$

Magnitude R_T represents the loss of effective power due to the presence of a water-ejecting pipe.

Using the experimental data, we find the extent of this loss for the different versions of model 331. To this end, we first define w_a by equation (10), and use the known magnitude v_s to calculate P_k by equation (14). We will now find the value ζ_T on the basis of (17) and (18).

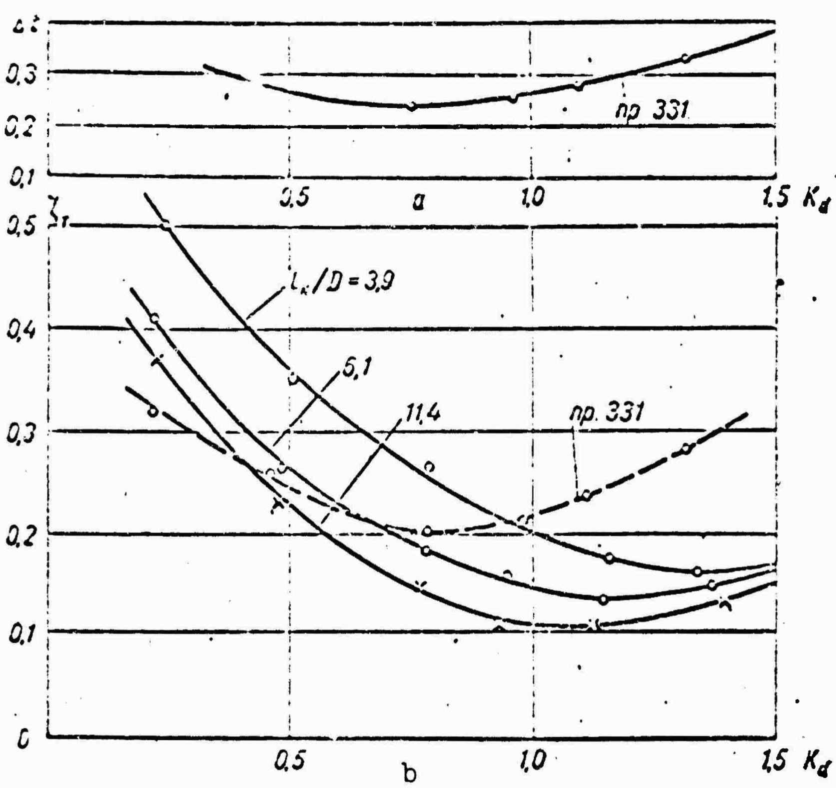

Figure 8. The $\zeta_T = f(K_d)$ magnitude of various alternate shapes of suction part of water duct.

The resulting data were used to plot the chart in figure 8b.

This figure (a) also shows the correction to the suction coefficient following the change from ideal to actual conditions. /78

Here

$$\Delta t = t_0 - t = \frac{p_k - p_c}{p}. \tag{19}$$

It is possible to conclude that magnitude ζ_T takes into account the full resistance of the water-spraying pipe produced by friction and, primarily, the shape of the water duct.

Actually, using the calculation formula to determine the loss by friction, as the water runs from the pressure motor through the pipe, we find that

$$\Delta h_T = \xi \frac{l}{D}\frac{v_s^2}{2g} \ [м] \qquad (20)$$

or

$$R'_T = \gamma \Delta h_T F = \xi \frac{l}{D}\rho\frac{v_s^2}{2}F \simeq 0{,}02\cdot 2{,}5\rho\frac{v_s^2}{2}F. \qquad (21)$$

It is assumed here that

$$\frac{l}{D} = 2{,}5; \quad \xi = 0{,}02 \qquad R_e = \frac{v_s D}{\nu} > 2\cdot 10^5.$$

It is obvious that the loss of effective power due to friction amounts to almost 15 percent of the entire R_T magnitude.

Furthermore, as investigation shows, magnitude ζ_T depends to a considerable extent on the shape of the water duct, the operating conditions (K_d) and, finally, in our opinion, on the scale effect on them. It should therefore be assumed that the support factor of the water-spraying pipe, proposed by the Leningrad Institute of Water Transport, is fairly /79
approximate.

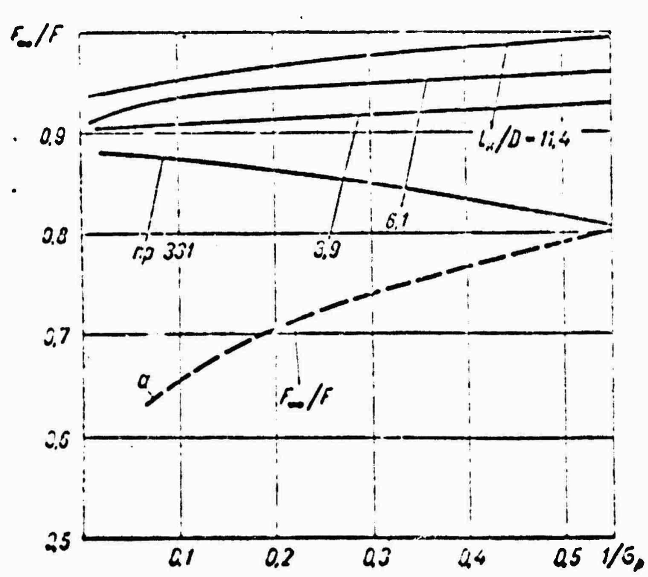

Figure 9. Evaluation of theoretical narrowing of water jet outside of engine.

To evaluate the magnitude ζ_T, when the shape of the suction part of the duct corresponds to the scheme shown in figure 1a, it is possible to use figure 8, where the dependence of ζ_T on l_k/D and the inverse load factor K_d is portrayed in graphic form.

Finally, we shall analyze the effect of the suction part of the duct on the shape of the water jet outside of the motor. Let us determine the following value

$$\frac{F_\infty}{F} = \frac{v_s}{v_\infty} = \frac{v_s}{v_e + w_a},$$

which characterizes the theoretical shape of the water jet as the ratio of its spray over a long distance F_∞ to the operational target F (fig. 9). In all cases the water jet is slightly compressed by the cylindrically shaped outlet part of the duct. A comparison of the curves $F_\infty/F = f(\sigma_p)$ with the theoretical curve of the insulated motor (curve a), for which $\frac{F_\infty}{F} = \frac{1+\sqrt{1+\sigma_p}}{2\sqrt{1+\sigma_p}}$, shows that the shape of the water jet, even with a high load coefficient, is close to cylindrical, as in the case of impellers with nozzles (ref. 7).

Some recommendations of the optimal shape of the spray nozzle can be found in the investigations of the Leningrad Institute of Water Transport carried out by E. I. Stepanyuk in 1960 (ref. 8).

CONCLUSIONS

1. A more reliable method of calculating the water-pressure motor, proposed by the Leningrad Institute of Water Transport, calls for a definition of the suction coefficient t and relative speed v/v_s in the ... of the propeller ... calculations of the adopted form of the suction part of the water-jet duct.

The tests already made make it possible to determine these parameters ... of the longitudinal extension of the duct with a direct intake.

2. It was found that an increase in the longitudinal opening l_k/D /80 has a positive effect on the propulsive capacity of the complex as a whole because of the reduced suction losses.

The shapes of the suction channel as outlined in figure 1a, may be proposed for high-speed boats (operating in small roadsteads).

In the case of tugboats it would also be expedient to increase the cross section.

3. A system of corrections for partial depths is proposed for a more precise definition of the suction factor, when the propeller is not at full depth ($T_B/D < 1$).

4. The diagrams outlining the values k_w and ζ_T make it possible to select these calculated parameters that take into account the characteristic shape of the suction part of the duct.

In general, the experimental data on the hull-engine interaction of a fireboat, the propeller speed and the losses in the water duct improve the method of calculating fireboat designs, proposed by the Leningrad Institute of Water Transport.

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